TM no. M-51-76-16

TECHNICAL MEMORANDUM

title: "DYNAMIC ANALYSIS OF THE P&H 6250 TRUCK CRANE EMPLOYING THE 150 FOOT BOOM STICK MODEL, by

author: Fr~~ank~~ R. Johnson, Jr.

date: Sep~~tember~~ 1976."

sponsor: NAVAL FACILITIES ENGINEERING COMMAND

program nos: YA1X6-001-01-001F

CIVIL ENGINEERING LABORATORY

NAVAL CONSTRUCTION BATTALION CENTER
Port Hueneme, California 93043

PREFACE

Operational Requirement (OR-YSL03) for the Container Off-Loading and Transfer System (COTS) addresses the need for an integrated cargo handling system for discharging non-self-sustaining container-capable ships and other ships and barges at open beach sites and identifies the Navy's responsibility for developing certain elements of the required overall cargo handling system. DOD policy is documented in the DOD Project Master Plan for Surface Container Supported Distribution and the DOD system definition paper ''Over-the-Shore Discharge of Containerships (OSDOC) System.''

The Navy's version of the container distribution elements constitute the Container Off-Loading and Transfer System (COTS). The COTS advanced development program includes (a) the ship unloading subsystem, (b) the ship-to-shore subsystem, and (c) system level elements. The ship unloading subsystem includes: (a) ship/barge candidates, (b) cranes, (c) crane integration with ships/barges and (d) moorings. This report addresses the progress and accomplishments associated with the crane element of the ship unloading subsystem.

CONTENTS

INTRODUCTION

The Civil Engineering Laboratory was requested by the Naval Facilities Engineering Command to develop an analytical procedure for determining performance limits for mobile cranes operating from floating platforms moored in the open sea. The mobile lift cranes commonly used in the construction industry being considered range from 300 to 600 ton rated capacity. The floating platform supporting the lift crane can be a seagoing barge or a ship that has been modified to support the crane on its deck.

There are four basic components that make up the lift crane according to the Power Crane and Shovel Association [1]. These components are: the mobile mounting, the revolving superstructure, the power plant, and the front end operating equipment. These components will be explained in further detail in ensuing paragraphs.

The mobile mounting provides a fixed foundation from which the crane can operate and it also provides a means to transport the machine. One of the two types of mobile mountings is the crawler mounting. This mounting has two continuous, parallel crawler belts that provide the forward and reverse mobility in much the same way as a common bulldozer. The second type of mobile mounting is rubber tire carrier mounting. The truck type of mounting usually has separate engines for the carrier and the superstructure, whereas the self-propelled mounting generally has a common engine for the carrier and superstructure. In addition to transportability the mobile mounting provides the basic operating foundation for the machine. This foundation is enhanced to withstand overturning by using outriggers extending from both sides of the carrier. Another type enhancement employs the use of a circular ring, or ringer, that encircles the carrier to increase the machine's stability.

The revolving superstructure, or simply superstructure is the rotating frame and the machinery supported by the rotating frame. The main hoist system, the boom hoist system, the swing system, and the gantry are the primary elements of the revolving superstructure. The main hoist system includes the machinery and controls necessary to raise and lower the service load. The boom hoist system consists of the machinery and controls to raise and lower the boom, or in other words, control the boom angle. The swing system involves the machinery and controls for the rotation of the rotating superstructure and, by so doing, the swing system controls the swing angle. The gantry or ''A'' frame is the point of attachment for the boom supports. Each of the components of the revolving superstructure contribute to the primary functions of the machine; however the power plant, also located on the rotating frame, is usually considered separately as one of the four basic crane components.

The basic machine can perform various types of crane and excavating functions. The attachments associated with the various functions are usually collectively referred to as the front end operating equipment. The lift crane function is of interest in the work described herein. The front end equipment for the lift crane includes the basic boom structure, the boom hoist ropes and suspension, and miscellaneous attachments such as the hook and spreaders. The basic boom structure consists of a lower or base section, an upper or tip section, and insert sections placed between the lower and upper sections. The boom is supported by the boom suspension system which also serves as the boom hoist ropes. There are three types of boom suspension arrangements; continuous suspension, pendent suspension, and mast suspension. The continuous suspension is characterized by reeving the boom hoist ropes between sheaves located near the top of the gantry and the top of the boom tip without interruption. The pendent suspension system has a sheave arrangement located between the sheaves on the gantry and the boom tip. This sheave arrangement is supported by stationary ropes or pendents attached to the boom tip, and the boom hoist ropes are reeved between the gantry sheaves and the sheaves supported by the pendents. The mast suspension consists of a strut or mast between the mast hinge point located at or near the boom foot and sheave arrangement located at an intermediate position between the boom tip and the gantry sheaves. The mast is supported by the boom hoist ropes between the gantry and mast sheaves and a set of pendent ropes between the mast and boom tips.

The overall objective, as stated in the first paragraph, is to determine the performance limits of lift cranes when they are used under the dynamic conditions imposed while operating from a floating platform moored in the open sea. There are portions of the four basic components previously described that contribute to the overall limitations on the machine when used under dynamic conditions. The results presented in this and subsequent reports will address the crane components one at a time until the weak components have been identified and the associated performance limits have been established. This particular report addresses the lift crane front end components.

OBJECTIVE

The objective of the work reported herein is to demonstrate an analytical procedure for establishing the boom and boom suspension performance limits when the lift crane is operating from a floating platform moored in the open sea. The demonstration is accomplished by presenting a load rating curve for a P&H 6250 truck crane (TC) using an 150 foot boom. The steps used to determine the performance limits are demonstrated employing a procedure that requires refinement and using simplified engineering assumptions. The secondary objective is to determine the conditions that significantly contribute to the boom and boom suspension performance limits of the operating machine.

APPROACH

The P&H 6250 TC was selected for the work addressed in the report. This selection was based upon the availability of the machine for future tests in the dynamic operating environment at sea. Furthermore, with the cooperation of the Harnischfeger Corporation of Milwaukee, Wisconsin, structural and mechanical details and the results of the SAE J987 test reports were available.

The list shown in Table 1 indicates the various items that are considered in the establishment of performance limits for lift cranes subjected to dynamic conditions. In order to simplify the analysis problem so various factors contributing to the analysis results can be isolated, only the items indicated in the table by an ‘‘X’’ are considered in this report.

The boom and supporting structure of the P&H 6250 were modeled in a simplified fashion. The finite element method of linear structural analysis was used to analyze the simplified model. With this method of analysis, one can investigate the static and dynamic stress and deflection properties of each of the lacing and chord members in the boom. However, an analysis involving this level of detail is costly. Therefore, a simpler model referred to as the ‘‘stick’’ model was used.

The stick model referring to Figure 1, is made up of a series of nine beam elements and one truss element. The beam elements are prismatic but each element has been developed using an average of the nonprismatic section geometry taken from the actual structure at the location of the element end points. Thus, the nonprismatic geometry has been approximated by a series of prismatic elements. The nine beam elements that represent the actual boom consider the chords and the reinforcing plates located at the boom tip and base, but the boom lacing members have been omitted. Bending and axial loads can be investigated with this assumption but torsion and shear are not addressed. The suspension system is considered as a truss element with end points located at the gantry tip and the pendent support locations on the boom tip. This arrangement approximates the continuous suspension system instead of the pendent support system actually used in the structure. The P&H 6250 TC has an 150-foot boom made up of the 70-foot basic boom (40-foot base section and 30-foot heavy duty tip section) with 30-foot and 50-foot inserts.

The floating platform used in this analysis was a Delong Type B Barge (FDL-B). The barge is 150 feet long, 60 feet wide and it has a 2.32 foot draft. The platform motions were obtained by using a modified version of the RELMO computer program [2]. The modification of RELMO involved removing the relative motion features from the program in order to obtain the acceleration time history for six degrees of freedom (three translation and three rotation) at the center of gravity of the platform. However, the analysis discussed in this report only used the three translational acceleration time histories.

Two coordinate systems were used in this study. Figure 1 shows the structure coordinate system and the platform coordinate system. The origin of the structure coordinate system is located on the ground level just below the tip of the extended gantry (the carrier is resting on the x-z plane). The origin of the platform coordinate system is located at the center of gravity of the platform. The analysis assumed the origin of the structure coordinate system and the platform coordinate system coincide. Provision was made to vary the boom angle, the angle between the boom and the x axis of the structure coordinate system (75, 65, 52, and 41 degrees were used). Furthermore, the orientation of the platform coordinate system with respect to the structure coordinate system was varied so the boom tip could be oriented either over the bow or over the side of the platform.

The static loads imposed on the structure include both dead and live loads. The dead load was applied considering consistant mass formulation to calculate the uniform dead load on each member. The live loads considered included the lifted load, the hook, the spreader, the weight of the hoist rope, the force in the hoist rope directed toward the center of the hoist drum, and the side load. The vector components of these loads were resolved so they would be parallel to the structure axis system. The side load, applied normal to the x-y plane or plane of the boom is taken as 2 percent of the vertical load component which is consistant with industry practice.

The dynamic loads imposed were limited to the inertial effects of the translational platform motion. The mass of the structure was lumped at each of the beam element node points, and the mass of the lifted load, the hook, the spreader, and the hoist ropes was lumped at the tip of the boom. The motion of the platform and the resulting translational acceleration was calculated for head, quartering, and beam seas and considering a sea state three with a 5-foot significant wave height.

The evaluation of the stress and deflection levels experienced was limited to normal stress and industry accepted deflection criteria. The allowable stress was chosen to be the yield stress; 100,000 psi for the boom material and 204,000 psi for the suspension rope. However, where required, the allowable stress was reduced in accordance with the Euler buckling curve. The allowed buckling stress was not permitted to exceed the proportional limit taken as 75,000 psi in order to limit the evaluation to the elastic range which is consistant with the linear analysis that was conducted. The out-of-plane deflection of the tip of the boom was not permitted to exceed 36 inches (24 in./100 feet of boom) as suggested by SAE J987 [3].

RESULTS

The analysis results presented in this section begin with a comparison of the frequencies associated with the various components of the model. The displacement and acceleration limits for the FDL-B barge are presented. The results of a parameter study are discussed to establish the contributions of the various components to the overall results. Finally, the analysis results will be presented and derating curves for various conditions will be compared.

The frequencies and associated periods for the waves impinging upon the barge, the roll of the barge, and the boom and suspension structure are shown in Table 2. The first and tenth modes are presented for the boom and suspension structure. It is observed that the natural frequency of the barge roll and of the boom structure are very close in magnitude.

The motion characteristics of the FDL-B barge are presented in Tables 3 and 4. It is observed after comparing the probable maximum value for 1,000 wave cycles [4] shown in Table 3 to the maximum values obtained from a single 200-second simulation that the motion characteristics used in this study are less then the expected maximum. Moreover, the maximum conditions occur when a beam sea impinges upon the barge.

The information presented in Table 5 summarizes the results of a parameter study to indicate the contribution of the various loading conditions toward the establishment of the performance limits of the crane. It is observed that the structure weight does not appreciably contribute to the suspension strength, the boom strength, or the out-of-plane tip deflection. However, the structure mass does contribute to the dynamic response of the structure in that the mass has some affect on the out-of-plane deflection. The vertical and horizontal vectors, comprised of the vector components of the lifted load, the hook, the spreader, the main hoist rope, and the force in the main hoist rope, have an appreciable effect on the suspension and boom strength. The 2 percent of the vertical load applied normal to the plane of the boom, representing a side load, contributes very little to the out-of-plane deflection and has almost negligible effect on the suspension and boom strengths. The direction of the impinging wave has little effect on the structure when the boom is over the side. This is also true when the boom is over the bow and the waves approach from either the bow or quarter. However, when the boom is over the bow and the waves approach from the beam the tip deflection becomes significant. If the mass of the vertical load is removed from the tip and the boom is placed over the bow with waves approaching from the beam, the boom strength is affected to a small extent. Moreover, when the dynamic analysis was conducted without the contribution of the static loads the suspension ropes experienced compressive loads. Since ropes cannot sustain compressive loads, the net effect when static conditions are considered is less tension.

The curves presented in Figure 2 show the results of the analysis for loads between 60 and 100 kips. Analysis results not shown in this report indicated that the structure was able to pick up loads that are less than or equal to 60 kips without regard to operating radius. Furthermore, for loads in excess of 100 kips, the structure exceeded permissible strength and deflection limits. The presentation in the Figure 2 considered the static and dynamic conditions, and it was observed that out-of-plane deflection governed in each case except for the 41 degree boom angle where boom strength governed. Figure 3 displays the results of the maximum permissible load with respect to the operating radius. Comparing this figure with curves taken from load rating charts for the P&H 6250 TC with the 150-foot boom equipped with the heavy duty tip indicated that boom strength, suspension strength, and tip deflection do not govern the selection of the permissible load. Recalling that the analysis reported here was conducted under the assumption that the machine could be tied down without regard to the overturning moment, the differences in the curves can be anticipated.

CONCLUSIONS

The results presented indicate that the dynamic conditions, the linear analysis limitations and acceptance criteria are of great importance. Each of these items will be discussed in detail.

The dynamic conditions imposed upon the structure have a pronounced influence upon its performance, especially the beam sea. The first item to be considered is the probable maximum time history amplitudes that may arise in a six to twelve month period. The effects of the platform rotation and the transformation required to permit locating the origin of the structure axis system at a location other than the origin of the platform axis system will produce even more pronounced dynamic results.

The use of linear analysis is desired, however possible short comings arise. Due to the limitation to small angle theory and the stability of the stiffness method, use of the pendulum to impose the side load is precluded. However, the assumed side load criteria (i.e., 2 percent of the vertical load) commonly used in static analysis does not seem to be realistic and applicable in a dynamic analysis. Furthermore, placing the mass of the lifted load at the tip of the structure significantly influences the analysis results. Performing the analysis with a nonlinear structural analysis program can permit consideration of these conditions in a more realistic manner. Afterwards, linear analysis techniques can be developed to simply describe the structural response, because the effects of nonlinear behavior on linearizing assumptions will be known.

The boom strength and the suspension strength criteria seem to be applicable to the problem being solved. However, the deflection criteria which governs in most cases considered should be validated for application to the dynamic problem. Furthermore, the considerations of tie down design criteria should be included because comparison with the previously published load radius data indicates that a marked difference exists between the results using boom strength and deflection criteria and the published data. Machine stability and the pedestal capacity account for the differences.

The assumption used in a previous investigation [5] which states in effect that the static conditions are sufficient to prevent compression in the boom suspension ropes is valid according to the results presented in this analysis.

RECOMMENDATIONS

The preceding conclusions indicate that the nonlinear analysis should be conducted to more adequately represent the out-of-plane load imposed by the load as it swings in and out of the plane of the boom. Furthermore, the nonlinear analysis will also permit studying the pendent suspension system which was not a part of the analysis reported herein. It is also pointed out that the nonlinear analysis will permit placement of the vertical load mass in its proper and more realistic place located at the end of the hoist line.

Greater attention should be directed to the base motion conditions. This will include the addition of the rotational components to the base motion input, and the transformation required to vary the location of the structure axis.

The forthcoming full scale tests should be aimed at obtaining data for use as input for the analysis procedure. The motion experienced at the base of the boom with respect to the center of gravity of the vessel is paramount. Furthermore, the load time history imposed at the boom tip is also important.

The test results should be used to improve the model and subsequently the analytical prediction of the structural response to the dynamic conditions imposed on the crane. Comparison of the test and analytical results will increase the confidence for the validity of the data presented in this report.

ACKNOWLEDGMENT

The assistance and direction of Dr. Carley C. Ward is deeply appreciated. Mr. Duane A. Davis has also assisted with the application of the platform motion calculations to the structural analysis problem.

REFERENCES

1. Power Crane and Shovel Association, ''Mobile Power Crane and Excavator Standards,'' PCSA Standard No. 1, Milwaukee, Wisconsin, 1968.

2. Davis, D. A., and Zwibel, H. S., ''The Motion of Floating Advanced Base Components in Shoal-Water – A Comparison Between Theory and Field Test Data,'' Civil Engineering Laboratory, Technical Note N-1371, Port Hueneme, CA Jan 1975.

3. Society of Automotive Engineers, ''Crane Structures – Method of Test – SAE J987.'' SAE REcommended Practice, New York, NY, Mar 1973.

4. Michel, W. H., ''Sea Spectra Simplified,'' Marine Technology, Vol 5, No. 1, pp 17-29, January 1968.

5. Saul, W. E., ''Computer Simulation of Ship Mounted Crane Booms to Wave Motion,'' The University of Wisconsin, Madison, Wisconsin, Nov 1974.

6. Harnischfeger Corporation, ''Standard Crane Ratings Over Side and Rear, PCSA Class 18-1775,'' P&H 6250 to chart 32R160, Milwaukie, Wisconsin.

7. Harnischfeger Corporation, ''300 Ton Truck Crane Over Rear Ratings With Mast,'' P&H 6250 TC chart B5-1073, Milwaukie, Wisconsin.

8. Harnischfeger Corporation, ''P&H 6250 TC Upper Pedistal Mounted With 41985 Kilograms Counterweight,'' P&H 6250 TC chart 32R250, Milwaukie, Wisconsin.

Table 1. Considerations for Establishing Lift Crane Dynamic Performance Limits [a]

Applied Loads				
	Dead Load			X
	Live Load			
		Vertical Component		X
		Out of Plane Component		X
		Off Lead Component		
		Hoist Line Component		X
		Fleet Angle Component		
	Wind Pressure			
		On the boom		
		On the Load		
	Dynamic Load			
		Platform drop out		
		Rebound		
		Colliding Objects		
		Luffing		
		Swing		
		Hoisting		
		Lifted Mass at Boom Tip		X
		Free Pendulation		
		Restrained Pendulation		
		Translational Base Motion		
			Beam Sea	X
			Quartering Sea	X
			Head Sea	X
		Rotational Base Motion		
			Beam Sea	
			Quartering Sea	
			Head Sea	
Boom Position				
	Over the Bow			X
	Over the Side			X
	Over the Quarter Point			
Boom Length				
	100 foot 40-10-20-30			
	110 foot 40-10-30-30			
	120 foot 40-20-30-30			
	130 foot 40-30-30-30			
	140 foot 40-20-50-30			
	150 foot 40-30-50-30			X
Boom Suspension				
	Continuous			X
	Pendent			
	Mast			

Table 1. continued

Analysis			
	Linear		X
	Nonlinear		
Evaluation			
	Threshold Stress		X
	Fatigue Endurance		
	Tip Deflection		X
	Overturning		
		Without tie downs	
		Infinite Tie Downs	X
		Designed Tie Downs	
	Machinery		

[a]The items identified by the X were considered in the analysis being reported.

Table 2. Period and Frequency Values for the Waves, Platform and Boom-Suspension Structure

	Waves[a]		Platform[b]	Boom[c]	
	Maximum	Minimum	Roll	Maximum	Minimum
Period (sec)	50.00	1.05	2.40	2.70	0.02
Frequency (cycles/sec)	0.95	0.02	0.42	49.28	0.37

[a] Range of wave periods assumed in the platform motion simulation.
[b] Platform natural roll period and frequency.
[c] Boom first and tenth mode shape period and frequencies.

Table 3. Probable Maximum Displacement Amplitude in 1,000 Cycles for the Floating Delong Type B Barge[a]

Wave Direction	Wave Height (ft)	Translation			Rotation		
		Surge (in.)	Sway (in.)	Heave (in.)	Roll (deg)	Pitch (deg)	Yaw (deg)
Head	5.0	12.7	0.0	15.5	0.0	3.0	0.0
Quartering	5.0	12.1	14.2	22.5	1.5	3.0	1.5
Beam	5.0	0.0	31.2	36.4	5.8	0.0	0.0

[a] Presented values are computed by multiplying 1.86 times the simulated significant motion values as suggested by Reference 4.

Table 4. Maximum Displacement and Acceleration Amplitudes for the Floating Delong Type B Barge[a]

Wave Direction	Wave Height (ft)	Translation: Surge (in.)	Translation: Sway (in.)	Translation: Heave (in.)	Rotation: Roll (deg)	Rotation: Pitch (deg)	Rotation: Yaw (deg)
		Displacement					
		(in.)			(deg)		
Head	5.0	10.9	0.0	12.1	0.0	2.2	0.0
Quartering	5.0	9.8	9.3	14.0	1.1	2.6	1.1
Beam	5.0	0.0	21.4	27.9	4.7	0.0	2.0
		Acceleration					
		(in./sec^2)			(deg/sec^2)		
Head	5.0	13.0	0.0	11.7	0.0	2.3	0.0
Quartering	5.0	8.8	8.6	12.3	1.3	2.3	1.2
Beam	5.0	0.0	42.2	30.5	18.1	0.0	0.0

[a]Obtained from a single 200 second motion simulation.

Table 5. Analysis Summary for the P&H 6250 TC, 150 Foot Boom at 75 Degrees, 5-Foot Significant Wave Height

Wave Direction	Boom Location	Load Components	Suspension[a] Strength Ratio	Boom[a] Strength Ratio	Tip[a] Deflection Ratio
N/A	N/A	dead Load	0.02	0.04	0.00
N/A	N/A	vertical load	0.23	0.25	0.00
N/A	N/A	side load	0.00	0.05	0.08
Beam	bow	static & tip mass	0.25	0.68	0.70
Beam	bow	tip mass	0.03	0.38	0.62
Beam	bow	no tip mass	0.00	0.17	0.03
Beam	side	tip mass	0.17	0.16	0.00
Beam	side	static & tip mass	0.39	0.44	0.08
Head	bow	tip mass	0.03	0.07	0.00
Head	side	tip mass	0.02	0.18	0.19
Quarter	bow	tip mass	0.02	0.10	0.14
Quarter	side	tip mass	0.03	0.11	0.14

[a]The ratios reported are computed by dividing the predicted value by the allowed value.

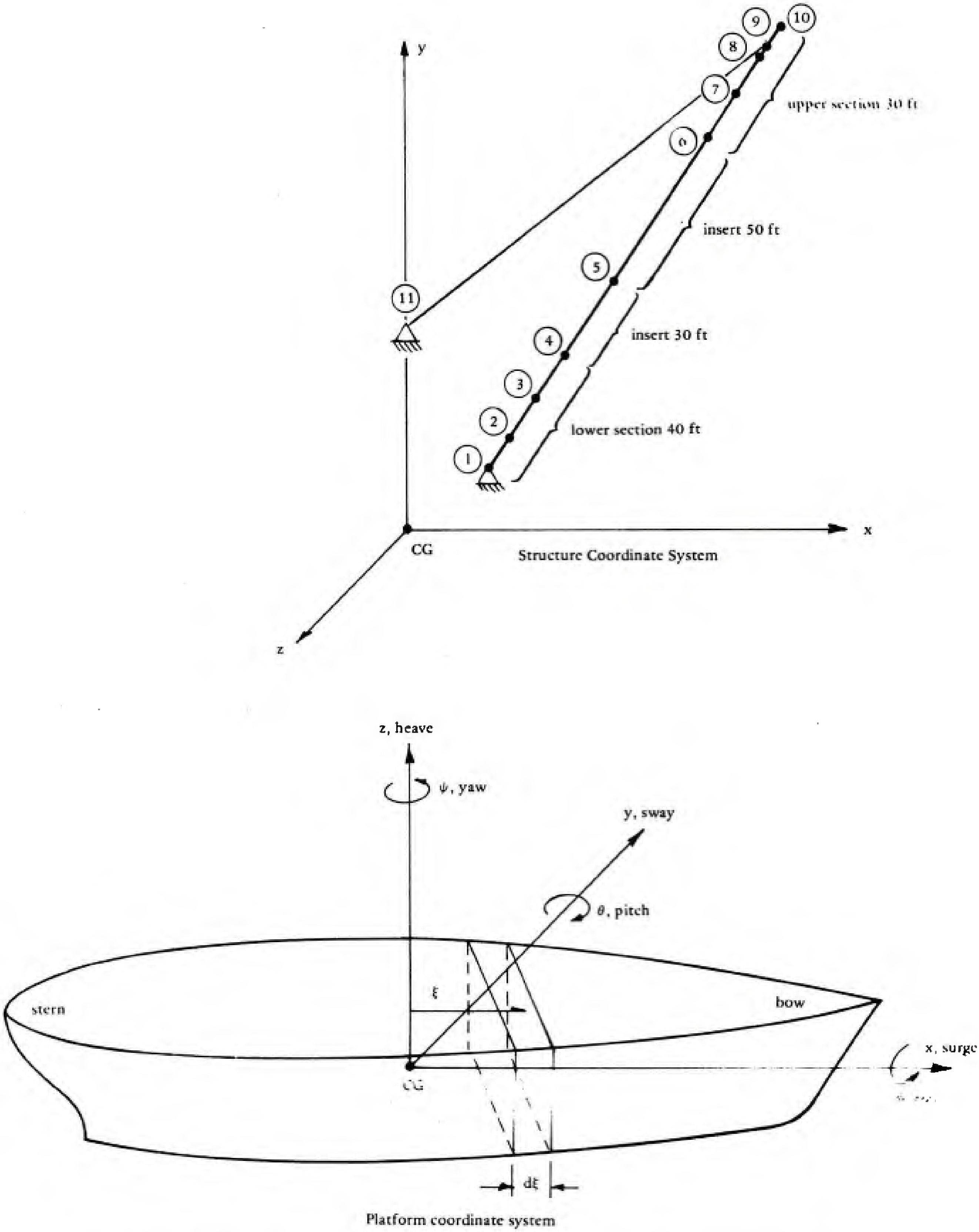


Figure 1. P&H 6250 TC model with respect to the structure and platform coordinate system.

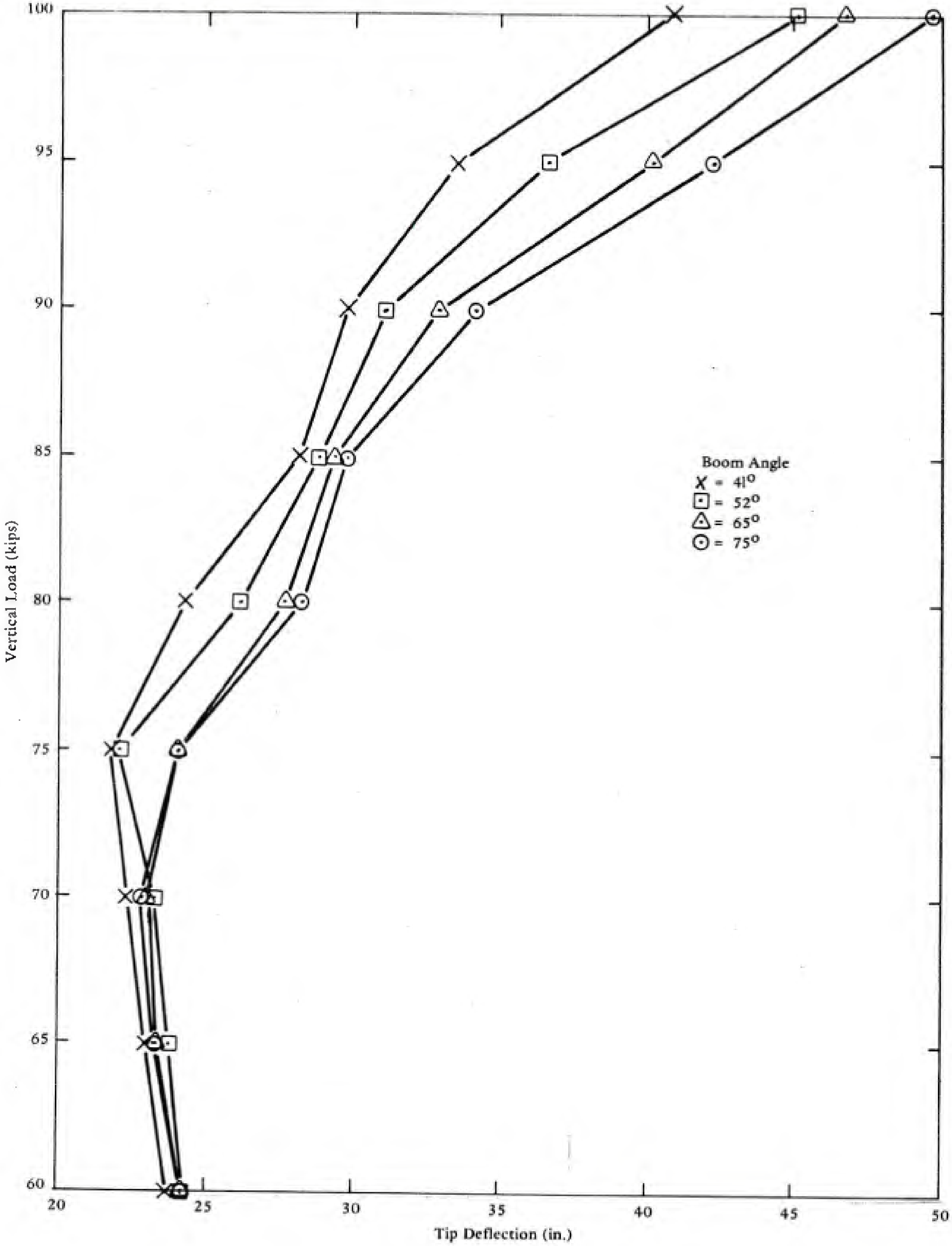
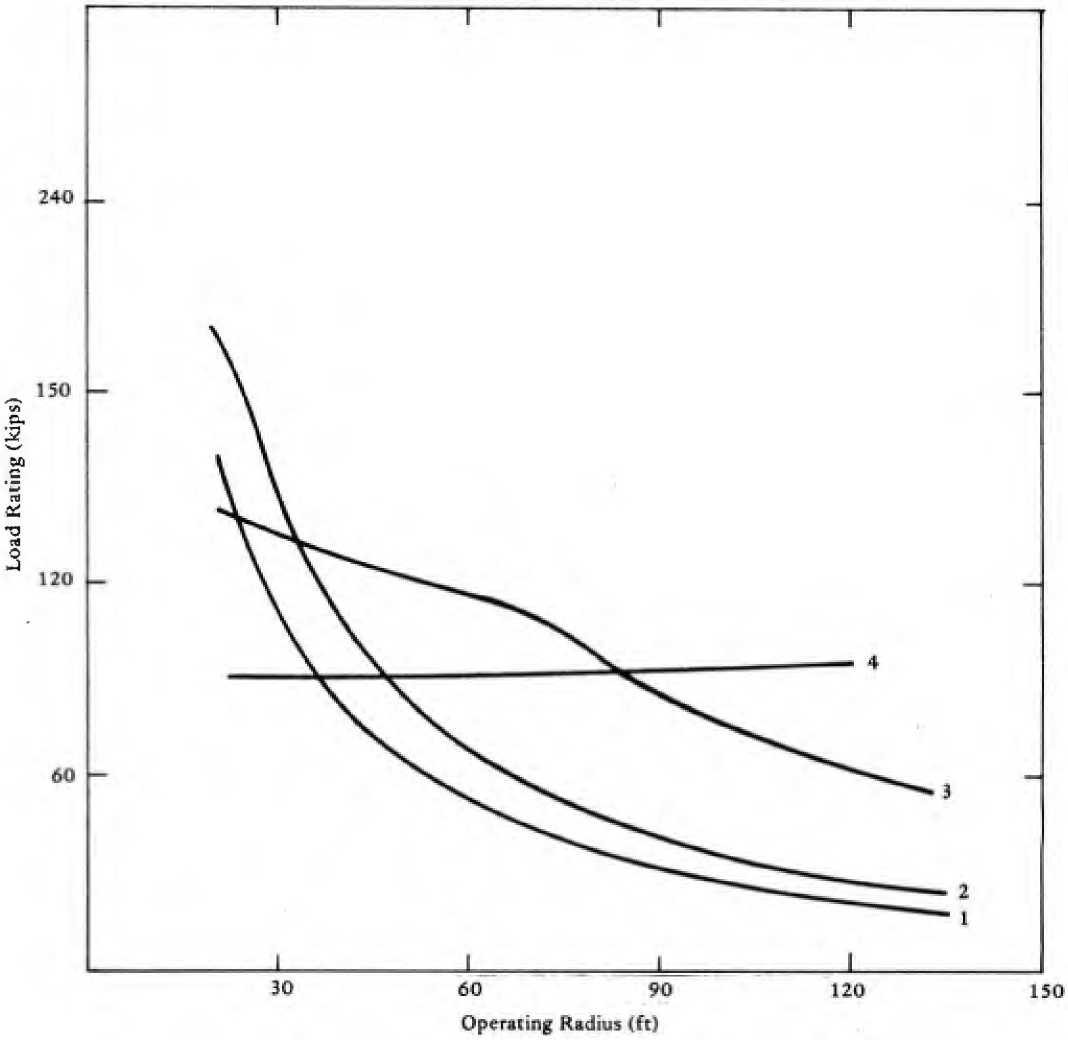


Figure 2. Vertical load versus tip deflection for P&H 6250 150-foot boom using continuous suspension, operating from a moored floating platform in a 5-foot beam sea.



Curve Number:

1. The heavy duty tip with pendent suspension and considering side load.[6]
2. The heavy duty tip with mast suspension and considering side load.[7]
3. The heavy duty tip with mast suspension, pedestal mounted on a barge (static analysis results).[8]
4. The heavy duty tip with continuous suspension, infinite tie down forces to withstand dynamic conditions imposed by a 5 foot significant wave impinging on the FDL-8 barge.

Figure 3. The rated load based on operating radius for a P&H 6250 TC with 150-foot boom equipped with a heavy duty tip.